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Development of a Cyclone-Vortex Nozzle Vane and Design Efficiency of Its Convective-Film Cooling by Steam

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Abstract—The effectiveness of using steam for cooling the first-stage nozzle vanes, one of the most thermally stressed components of gas turbines, is analyzed. Results obtained from calculation of the designed vane furnished with a cyclone-vortex cooling system carried out in the ANSYS CFX software package are presented. It is shown that use of steam in combination with flow swirling in the cooling system channels makes it possible to achieve essential enhancement of heat-transfer processes and better efficiency of cooling averaged over the vane profile.

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Improvement of the main performance indicators of gas turbine units (GTUs) is connected with a steady growth of gas temperature at the turbine inlet, which is achieved by applying new heat-resistant alloys and improving turbine cooling systems. This also involves an increase in the compressor's compression ratio. Power-generating GTUs with a temperature of gases as high as 1773 K and efficiency of 39–45% are now in operation. Almost all of them, including the most economically efficient aircraft-derivative GTUs with a compression ratio of up to 40 and power capacity of up to 100 MW have convective-film air cooling. The existing combined-cycle power plants (CCPs) with an efficiency of not less than 57% have been constructed on the basis of 2 to 3 times larger single-shaft GTUs [1] with not only air, but also steam cooling. A closed single-loop steam cooling system is used, e.g., for cooling the nozzles and the first- and second stages of the 9H gas turbine unit of General Electric. The 109H combined-cycle plant with a capacity of 520 MW constructed on the basis of this GTU has an efficiency of 60% [2].

With air and steam cooling systems being developed in parallel with each other, a tendency of making the maximal use of their positive properties for constructing a high-temperature CCP has emerged. By using steam generated in the GTU's heat-recovery boiler for cooling the turbine [3, 4], it is possible to achieve the same efficiency of domestically produced CCPs as that of those produced outside of Russia [5, 6].

The required efficiency of GTU cooling by air or steam supplied at a limited flowrate is achieved to a considerable degree by enhancing heat transfer in the channels of nozzles and blades through the use of various methods. The modified method of cyclone enhancement, in which air is admitted into a cylindrical channel through a few slots distributed over its length, is one of these methods. Two- and three-channel cyclone cooling chambers with cylindrical channels for inner cooling of blades have gained predominant development in research works [7]. As is seen from Fig. 1, heat transfer is enhanced most significantly in such channels. The Reynolds number, the value of which is determined from the channel's equivalent diameter and bulk flow velocity, was varied in the range $\mathrm{Re}_d = 8600–80000$. The data on enhancement of heat transfer in a vortex matrix, which are depicted in Fig. 1, were obtained at the values of Reynolds number ranging from 2500 to 10000. In this study, we proposed the basic design of a cyclone-vortex nozzle vane and carried out a calculation study of the efficiency of its steam cooling organizion via an open-circuit scheme and under the conditions of the high-pressure turbine used in a medium-capacity GTU.

The vane cooling arrangement is a system of cyclone tubes, coolant to which is admitted through tangential inlets from the central cavity, and a vortex matrix located in the vane's exit edge zone. The flow of coolant is organized in the following ways (Fig. 2):

(*a*) Coolant is supplied from the end-face inlet to the central cavity.

(*b*) Cyclone flow of coolant is organized by supplying and swirling it through tangential holes connecting the central cavity to peripheral cylindrical channels.

(*c*) Coolant is released from the leading edge cooling channel into the flow path through four rows of perforation.

(*d*) Collection of coolant from the peripheral channels and its supply to a vortex matrix are organized in a header located above the vane.

(*e*) Coolant leaving the vortex matrix is discharged into the flow path through a slot in the vane's trailing edge.

The vane's profile was designed taking into account that the steam cooling the vane is blown off into the flow path. The geometry of the cyclone steam cooling

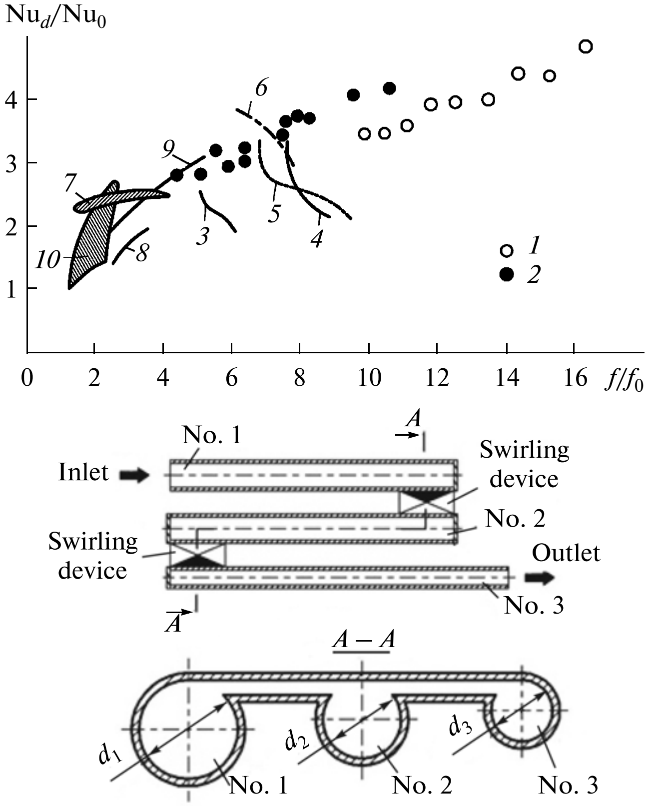


Fig. 1. Comparison of methods of heat transfer enhancement. Nu_d and f, and Nu_0 and f_0 are the Nusselt number and pressure drop coefficient in the studied and smooth channels, respectively [5]. The three-channel cyclone system: (*1*) channel No. 2, (*2*) channel No. 3, (*3*)–(*6*) straight and inclined (cut) ribs (60 and 90°), (*7*) and (*10*) surface indents, (*8*) pins, and (*9*) vortex matrix.

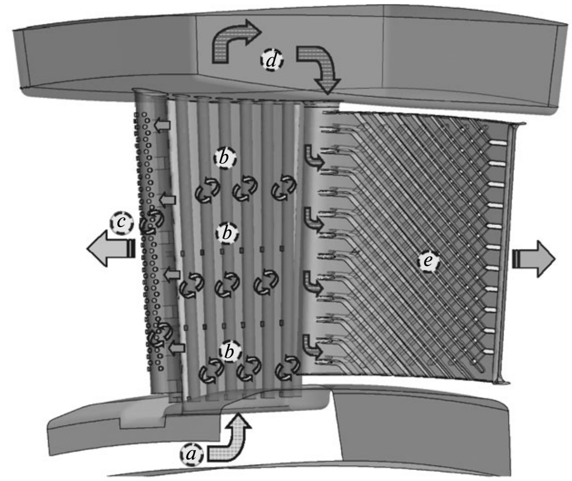


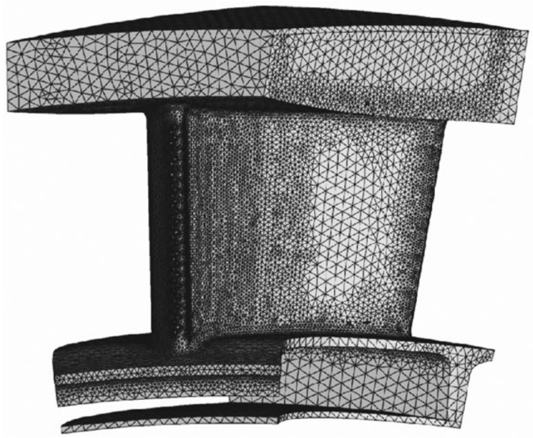
Fig. 2. Cyclone-vortex nozzle vane cooling system (the arrows conditionally show the flow of coolant).

system of the nozzle vane was determined using the well-known methods of calculating gas turbine cooling systems [8, 9] and a cyclone system for inner cooling of turbomachine blades [7]. The efficiency of the developed nozzle vane cooling system was determined by numerically simulating heat transfer and gas dynamics of the coolant flow in the inner channels taking into account the external stream of high-temperature gases flowing over the vane profile.

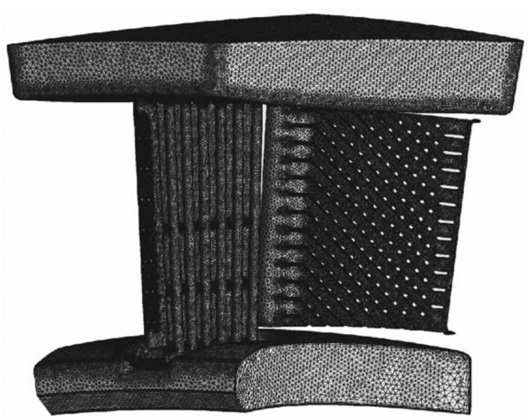
The constructed calculation region consists of three main subregions: the vane, the intervane channel, and the vane's inner cavity. The geometry of the intervane channel subregion was essentially a periodic sector. The flow path at the inlet to the intervane channel and at the outlet from it was extended by one and one-and-a-half vane's chord, respectively. Each subregion was quantized by suitable computational grids: tetragonal for the vane body, hexagonal for the intervane channel, and combined (comprising tetragonal and prismatic elements) for the inner cavity. The computational grid was constructed using the ANSYS ICEM CFD software package in compliance with the following requirements for boundary layer quantization: the first cell's height is equal to 5 × 10–6 m, the ratio between the volumes of adjacent cells is not more than 1.3, and the number of prismatic layers is equal to 10. The computational grid is shown in Fig. 3. Prior to carrying out detailed 3D calculations, we had analyzed literature sources [10, 11] devoted to simulation of conjugate heat transfer in turbine nozzle vanes in order to determine the most adequate calculation model. It was found from this analysis that the most trustworthy results on heat transfer from the gas flow to a vane's external surface are achieved when the SST k–ω turbulence model is used. The distributions of static pressure, wall temperature, and heat transfer coefficient were analyzed, and it has been found from this analysis that the error of calculation carried out using the SST k–ω model is around 8.8% in the leading edge zone and 5% in the middle sections of the vane's pressure and suction sides. The validity of applying the turbulence model for the inner cyclone channels was tested against the experimental data of [4], and its use has shown sufficient accuracy in calculating heat transfer from swirl flow of steam; the coefficients of heat transfer from the swirl flow were determined with an error of not higher than 8%.

To assess the quality of quantization used in the computational grid, we carried out a series of calculations for determining the discharge capacity of the model vane's cooling path under isothermal conditions, the results of which were used for carrying out verification against experimental data. Experimental hydraulic tests were carried out on the vane's stereolithographic model made of material on the basis of phenol-formaldehyde resins on a 1 : 1 scale (Fig. 4). These hydraulic tests made it possible to establish the dependence of air flowrate through the cooling system on the full pressure at the inlet.

(a)



(b)



(c)

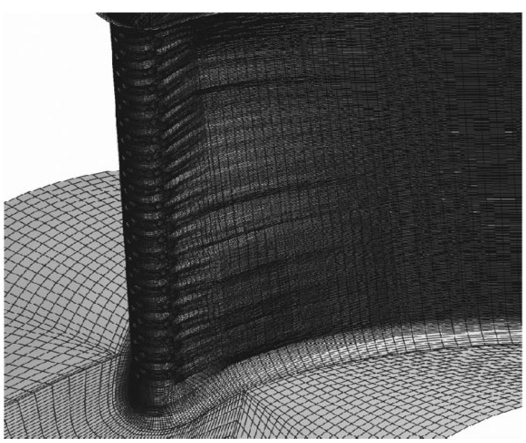


Fig. 3. Computational grids for the vane body (a), inner cavity (b), and interblade channel (c).

A fixed static pressure over the intervane channel (at the outlet from the perforation holes) was taken in calculations as a boundary condition. The full pressure at the system inlet was varied from 1.048×10^5 to 1.726×10^5 Pa. The absolute temperature at the inlet

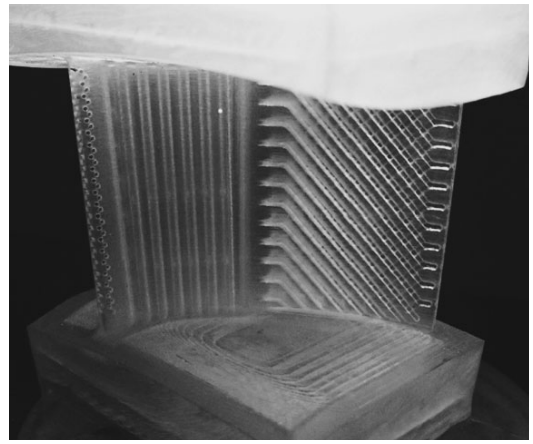


Fig. 4. Stereolithographic model of a cyclone nozzle vane.

was assumed equal to 297.35 K. The system of Reynolds equation was closed by the SST k–ω turbulence model. Additionally, it was assumed that the air viscosity and heat conductivity coefficients (Pa s) and [W/(m K)] can be calculated from the Sutherland formula [12]

$$\mu = \frac{1.45774\times10^{-6}T^{-1.5}}{110.4+T}; \tag{1}$$

$$\lambda = \frac{0.00250139\,T^{1.5}}{194.4+T}. \tag{2}$$

The dependence of isobaric specific heat of air [J/(kg K)] on temperature was given by the following polynomial of fifth degree:

$$\begin{aligned} c_p &= 1041.82976 + (-0.3467815)T \\ &+ (9.40237\times10^{-4})T^{-2} + (-6.66096\times10^{-7})T^3 \\ &+ (1.85509\times10^{-10})T^4 + (-1.28288\times10^{-14})T^5. \end{aligned} \tag{3}$$

The results of experimental operations for blowing through the stereolithographic model of a vane furnished with a cyclone cooling system and their comparison with calculation results are shown in Fig. 5. The calculated and experimental data were compared in terms of reduced flowrate

$$G_{\rm red} = G_{\rm phys}\sqrt{T^*}/p^*, \tag{4}$$

where T^* and p^* are the full temperature and full pressure at the cooling channel inlet.

The maximal deviation is observed for the pressure ratio $\pi = 1.726$ and makes up 9%. The verification that was carried out allows a conclusion to be drawn that the calculation and grid models used in the study are adequate for calculating the flow in the turbine vane cooling channels.

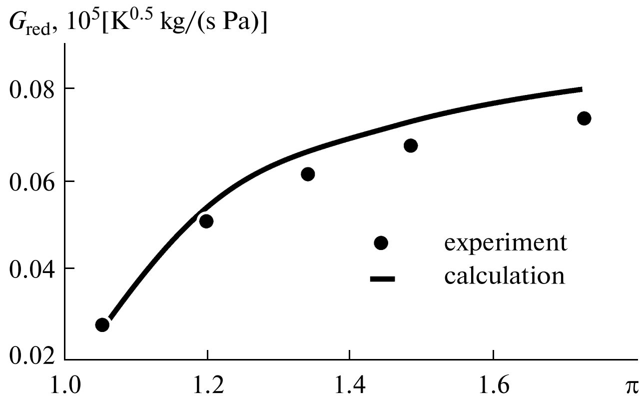


Fig. 5. Throughput capacity of the model vane's cooling path.

For estimating the thermal state of the vane during its operation in the working mode, we specified the following boundary conditions: the profiles of full pressure and full temperature during absolute motion as functions of channel radius at the inlet to the intervane channel; the value of static pressure at the outlet from the intervane channel, which was also obtained from the turbine calculation carried out on the assumption of ideal gas flow; and the values of full temperature and full pressure at the cooling system inlet. The properties of vane material corresponded to a ChS104 heat-resistant alloy. The values of all boundary conditions are given below.

Mean pressure at the inlet to the intervane channel p^*_{1m}, MPa	1.7283
Mean temperature at the inlet to the intervane channel T^*_{1m}, K	1773
Flowrate at the inlet to the cooling steam admission cavity $\overline{G}$, %	3.7
Temperature at the inlet to the cooling steam admission cavity T^*_2, K	673
Averaged static pressure at the outlet from the intervane channel p_{out}, MPa	1.3248

The values of viscosity μ, heat conductivity λ, and Prandtl number Pr of working gases flowing over the turbine nozzles were determined taking into account admixture of steam from the holes in the leading and trailing edges. The concentration of steam in the combustion products over the profile's contour was a variable quantity. The average fraction of steam over the outlet cross-section was equal to 4.46%. The initial quality of steam in the combustion products upstream of the nozzle row due to the standard content of moisture in atmospheric air and generation of steam during oxidation of fuel hydrogen in the combustion chamber

Distribution of flowrates among the vane cooling channels

Section	G_{phys}, kg/s	G_{phys}/G_{cool}, %	Section	G_{phys}, kg/s	G_{phys}/G_{cool}, %
Perforation 1	0.00321	9.14	Tube 7	0.00090	2.59
Perforation 2	0.00462	13.20	Tube 8	0.00091	2.61
Perforation 3	0.00751	21.43	Tube 9	0.00094	2.69
Perforation 4	0.00423	12.26	Tube 10	0.00092	2.63
Tube 1	0.00080	2.29	Tube 11	0.00096	2.74
Tube 2	0.00092	2.63	Tube 12	0.00094	2.69
Tube 3	0.00091	2.62	Tube 13	0.00096	2.74
Tube 4	0.00098	2.80	Tube 14	0.00091	2.60
Tube 5	0.00111	3.17	Trailing edge channel	0.00151	4.29
Tube 6	0.00114	3.26	Coolant discharge into the flow path	0.011	31.43

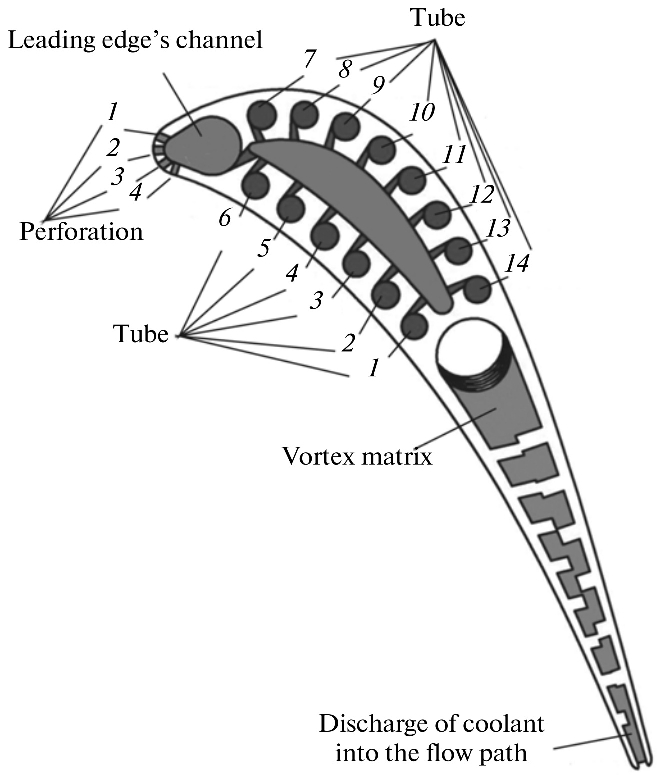


Fig. 6. Schematic notation of the vane's inner channels.

makes up 0.76%. The thermal–physical properties of the mixture of combustion products and steam were calculated in accordance with the model of ideal gas mixture laid down in the ANSYS CFX software package. The diffusion equation was additionally solved during this calculation, and the thermal–physical properties of the mixture were determined through the mass fractions and individual properties of mixture components.

The viscosity and heat conductivity coefficients of the mixture of air and combustion products were calculated using the Sutherland formula. The isobaric specific heat [J/(kg K)] was specified for the mixture as the following function of static temperature:

$$c_p = 957.17 + (0.2153)T + (5\times10^{-5})T^2 + (-3\times10^{-8})T^3. \quad (5)$$

The properties of steam (the viscosity, heat conductivity, and heat capacity coefficients) were specified on the basis of the international standard IAPWS IF 97 [13]. The calculations were terminated when the convergence in terms of normalized rms mismatches reached a level below 5×10^{-5} and when the imbalance of the total flowrate became less than 0.01%.

The table gives data on the distribution of flowrates over the vane's inner cavities, and Fig. 6 schematically shows the notation of the vane inner channels. It should be pointed out that the coolant flowrate has a sufficiently uniform distribution among the cyclone tubes, which will have a favorable effect on the distribution of heat flux removed by the cooling system. This fact points to the possibility of successfully using cyclone tubes in designing turbine blade cooling systems and arranging up a uniform temperature field on their surface.

The calculated distribution of the heat-transfer coefficient over the profile's contour at the specified

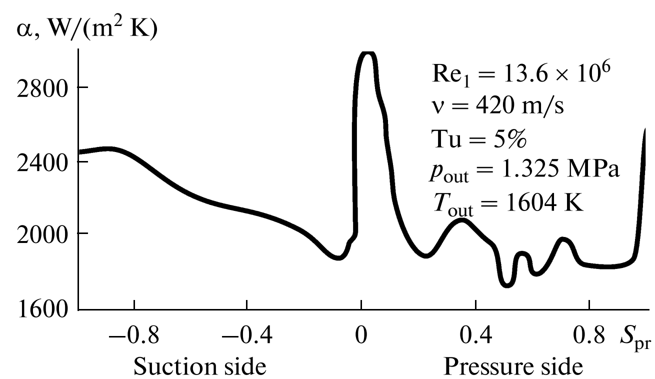


Fig. 7. Distribution of the heat transfer coefficient over the profile contour.

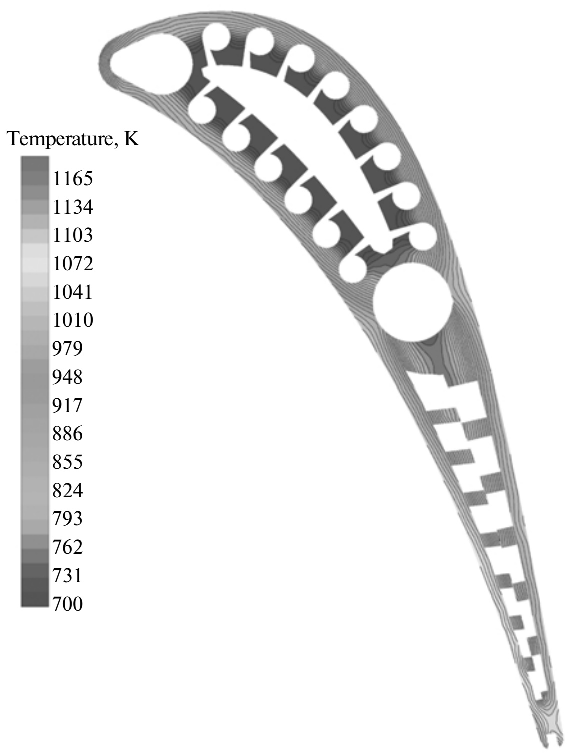


Fig. 8. Distribution of temperature in the vane body and over its surface.

Fig. 9. Coolant flow pattern in the vane's inner cavity.

turbulence intensity of incident flow (Tu = 5%) is shown in Fig. 7.

Figure 8 shows the distribution of temperature in the vane body for the middle cross-section, and Fig. 9 shows the coolant flow pattern in its inner cavity. The average temperature of the vane's outer surface is equal to 992 K, and its maximum value equal to 1135 K is reached near the leading edge. The vane temperature on the trailing edge surface does not exceed 1098 K. When coolant is supplied into the radial cylindrical channels tangentially, stable vortex structures are generated in these channels, which have an active influence on the boundary layer thus enhancing heat transfer processes.

It should be pointed out that the vane geometry was designed taking into account the intensive film cooling of the leading edge, because the strong thermal impact of the incidence gas flow at the stagnation and splitting point cannot be compensated only by inner convective cooling. In view of design considerations, as well as considerable flowrate of coolant blown out as a film curtain through the perforation holes, the cyclone channel of the entrance edge has a larger cross-section. This generated the need to arrange tangential admission of steam distributed over the height (from eight slots), which made it possible to prevent flow swirling from decaying and to maintain high values of heat-transfer coefficient in the inner channel over the entire height of the leading edge.

A uniform temperature field on the vane's leading edge is obtained by arranging the perforation holes in four rows (at 30° and 60° in both sides from the flow deceleration point) in a staggered order at 90° to the profile surface [14]. With such a configuration, the averaged parameters of steam injection (the ratio of

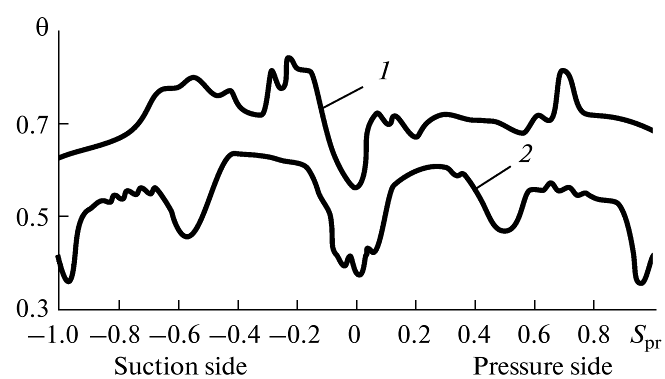


Fig. 10. Distribution of nozzle vane cooling efficiency over the vane's contour. (*1*) Steam cooling ($\theta_{mdl.sect} = 0.71$ and $\bar{G}_{cool} = 3.7\%$), and (*2*) air cooling ($\theta_{mdl.sect} = 0.53$ and $\bar{G}_{cool} = 3.7\%$).

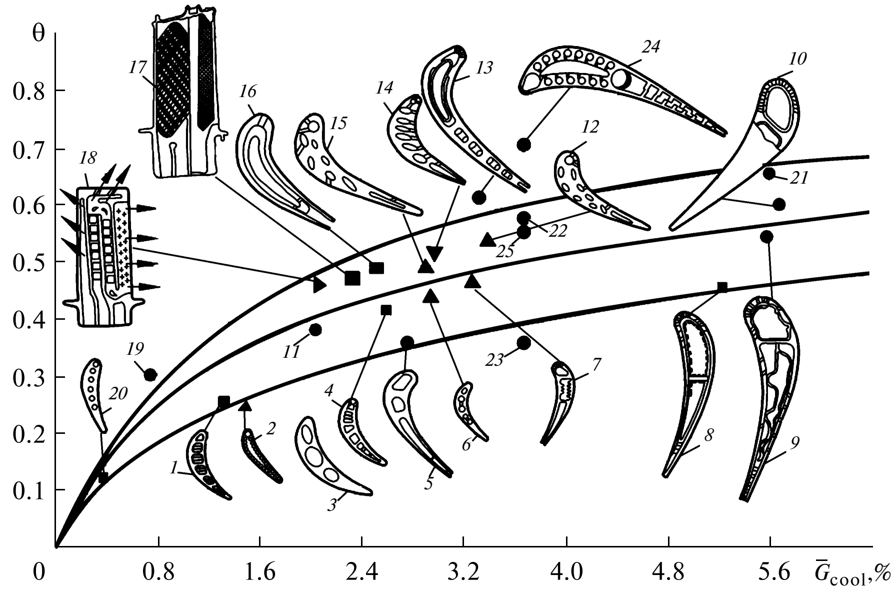


Fig. 11. Average cooling efficiencies of vanes with different configurations. (*1*) Blade with transverse cylindrical finning; (*2*) and (*8*) blades with small radial channels, (*3*) three-channel vane with loop-wise flow of coolant; (*4*) advanced vane *1* with blowing out coolant through the leading edge, (*5*) three-channel vane with release of coolant into the radial gap and trailing edge; (*6*), (*11*), (*19*), and (*20*) deflectorless perforated vanes with combined cooling; (*7*)–(*10*), (*13*), (*16*), and (*21*) deflector vanes; (*12*), (*14*), and (*15*) multichannel perforated vanes; (*17*) vane with a vortex matrix; (*18*) multichannel looped vane cooling arrangement; (*22*) and (*23*) mean and minimal cooling efficiency of an air-cooled cyclone-vortex nozzle vane; and (*24*) and (*25*) the same for a steam-cooled nozzle vane.

the densities of cooling air and incident gas flow) through the first, second, third, and fourth perforation rows were equal to 1.34, 1.46, 1.58, and 1.36, respectively. Approximately 57% of the entire flowrate of coolant passing through the vane is blown through the leading edge perforation.

The cooling efficiency was quantitatively estimated by the dimensionless cooling depth

$$\theta = \frac{T_g^* - T_v}{T_g^* - T_0^*}, \quad (6)$$

where T_g^* is the temperature of gas flowing over the nozzle vane (a mixture of air and combustion products), T_v is the vane temperature, and T_0^* is the initial temperature of coolant (steam).

Figure 10 shows the distribution of dimensionless cooling depth over the external profile in the mid-height cross-section. A comparative analysis of the air and steam cooling systems has shown that, given the same flowrate of coolant at the nozzle vane inlet $\bar{G}_{cool} = 3.7\%$, steam cooling is more efficient than air cooling by $\Delta\theta_{av} = 0.18$.

Nozzle vanes furnished with a cyclone-vortex cooling system have more efficient cooling as compared with vanes furnished with purely convective or convective-film cooling. This can be seen from the diagram shown in Fig. 11, which depicts the average cooling efficiencies θ of different vanes obtained under their working conditions vs. the relative flowrate of coolant $\bar{G}$. The curves shown in Fig. 11 represent the theoretical characteristics of cooled vanes constructed from the equation of heat balance.

In conclusion, it should be emphasized that, by arranging a cyclone-vortex (with flow swirling) system of cooling with steam, it is possible to achieve uniform distribution of temperature over the blade profile and height, which ensures high thermal efficiency of vane cooling $\theta = 0.71$ at the design temperature of gases equal to 1773 K and relative coolant flowrate equal to 3.7%.

REFERENCES

1. S. V. Tsanev, V. D. Burov, and A. N. Remezov, *Gas Turbine and Combined-Cycle Units for Thermal Power Stations* (MEI, Moscow, 2002) [in Russian].
2. E. John and H. Pritchard, "System^{TM} Technology Update," in *Proceedings of ASME Turbo Expo. Power for Land, Sea, and Air. Atlanta, Georgia, the United States, June 16–19, 2003.*
3. L. V. Arsen'ev, Yu. G. Korsov, and I. B. Mityaev, "Results from a Study for Analyzing the Efficiency of Cooling Gas Turbine Rotor Blades by Air and Steam," Trudy TsKTI, issue 165, 3–9 (1978).
4. L. V. Arsen'ev, I. B. Mityaev, and V. G. Polishchuk, "An Experimental Installations for Studying Heat Transfer in Rotor Blades," Izv. Vyssh. Uchebn. Zaved., Energetika, 151–152 (1975).

5. O. N. Favorskii, A. I. Leont'ev, V. A. Fedorov, and O. O. Mil'man, "Efficient Technologies for Producing Electric Power and Heat Using Organic Fuels," Teploenergetika, No. 9, 19–21 (2003) [Therm. Eng., No. 9 (2003)].

6. S. V. Veretennikov, S. M. Khasanov, and S. A. Smirnov, "A Study of Heat Transfer Enhancement in Cooled Blades of High-Temperature Steam and Gas Turbines by Flow Swirling," in *Proceedings of the Third International Conference "Heat-and-Mass Transfer and Hydrodynamics in Swirl Flows,"* (MEI, Moscow, 2008), pp. 1–8.

7. A. A. Khalatov, "Heat Transfer and Hydrodynamics in Fields of Centrifugal Mass Forces," in *Vortex Technologies of Aerodynamics in Construction of Power-Generating Gas Turbines* (Ukr. Nat. Akad. Nauk, Kiev, 2008), Vol. 7.

8. V. I. Lokai, M. N. Bodunov, V. V. Zhuikov, and A. V. Shchukin, *Heat Transfer in Cooled Parts of Aircraft Gas Turbine Engines* (Mashinostroenie, Moscow, 1985) [in Russian].

9. L. M. Zysina–Molozhen, L. V. Zysin, and M. P. Polyak, *Heat Transfer in Turbomachines* (Mashinostroenie, Leningrad, 1974) [in Russian].

10. G. A. Ledezmal, G. M. Laskowski, and A. K. Tolpadi, "Turbulence Model Assessment for Conjugate Heat Transfer in a High Pressure Turbine Vane Model. GT2008-50498," in *Proceedings of ASME Turbo Expo2008: Power for Land, Sea and Air, Berlin, 2008.*

11. D. Charbonnier, P. Ott, M. Jonsson, Th. Köbke, and F. Cottier, "Comparison of Numerical Investigations with Measured Heat Transfer Performance of a Film Cooled Turbine Vane. GT2008-50623," in *Proceedings of ASME Turbo Expo2008: Power for Land, Sea and Air, Berlin, 2008.*

12. W. Sutherland, "The Viscosity of Gas and Molecular Force," Philos. Mag., No. 5, 507–531 (1983).

13. W. Wagner and A. Kruse, *The Industrial Standard IAPWS-IF97: Properties of Water and Steam* (Springer, Berlin, 1998).

14. Yu. G. Gorelov, *Efficient Methods for Cooling Turbine Blades Used in Gas Turbine Units and Aircraft Gas Turbine Engines* (RGATA, Rybinsk, 2004) [in Russian].